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THERMODYNAMIC INVESTIGATION OF GAS TURBINES WITH CONTROLLED CONSTANT VOLUME COMBUSTION

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16. Abstract

A novel process is described, based on controlled combustion at constant volume. The efficiency and specific output are calculated, as are the pressures, temperature and compositions of the gas mixture obtained in different parts of the installation, and comparisons are made with values found for constant pressure combustion. Assuming the permissible blade temperature is the same for both processes, the achievable efficiencies and specific power outputs are compared, as a function of the compression ratio. The effect of different turbine efficiencies is considered.

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EXPERIMENTS WITH CONSTANT VOLUME COMBUSTION

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The development of gas turbines started with turbines operating according to the constant volume combustion method. Before this, experiments were carried out with constant pressure gas turbines, but these failed because of the low efficiencies which were achievable with the then prevailing technology. In order to avoid these difficulties, Holzwarth [1] focused his attention on the explosion turbine. This turbine variation did not become successful because of the considerable construction complexity and the large performance weight produced by it. Compared with the combustion motor it did not bring any substantial advantages. It was only because of the construction of gas turbines with constant pressure combustion that it was possible to develop high power engines with small performance weights, so that this combustion method is being used exclusively today.

From the thermodynamic point of view, the constant volume combustion method has the advantage over the constant pressure

* Numbers in margin indicate pagination in original foreign text.

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combustion method that the work expended per unit mass is greater and therefore the process efficiency is more favorable for the same amount of supplied energy, if the same initial conditions are assumed for the process.

Therefore, numerous attempts were made to exploit the performance and wear advantages of constant volume combustion [2-5, 8]. Combustion is then intermittent, i.e., the combustion chamber is completely or partly closed off by means of valve-like mechanisms during combustion. In this way, considerable throttling losses occur as well as long times for the gas change. Also there is a great susceptibility to disturbances because of the high rotation rates. Therefore, it has been suggested to use rotating components for controlling the combustion chambers arranged in the circumferential direction, or to locate the combustion chamber in a cavity within the rotor. The expansion is then done usually in the collection space, which is maintained approximately at the same pressure. Also it has been suggested that the combustion process be carried out according to the constant volume method and to drive several independent turbine stages or turbine wheels at various pressure ranges, by appropriate valve control. Other suggestions attempt to supply the gas intermittently to the first turbine stage within the pressure range above the charge pressure using a control. The combustion gas is continuously fed into a second turbine stage to a collection space at a constant pressure on the order of the charge pressure.

In the following we make a thermodynamic investigation of a gas turbine process suggested by F. A. F. Schmidt [3], in which the combustion chamber operates intermittently and is controlled by rotating disks, and which have advantages compared with the previously-known or suggested working methods .

THE COMBUSTION CHAMBER

Method of Operation

Figure 1 shows a diagram of the combustion chamber. The combustion is introduced in the primary chamber a, which is closed off by means of two rotating disks b and c during combustion, and the seal between the chamber and the disks insures that only a small amount of gas flows out and therefore the pressure produced by combustion does not drop off noticeably.

In this way one obtains approximately a constant volume combustion. The gases, which are generally only partly burned, go from the primary combustion chamber with a great deal of vorticity into the secondary combustion chamber d, in which post-combustion and pressure equalization occurs. From here on, the combustion gas expands in the turbine or in a thrust nozzle. The inlet and outlet control organs of the secondary combustion chamber are also in the form of rotating disks (b and e). Because of the inertia of the gas mass flowing out of the primary part of the combustion chamber into the secondary part, an underpressure can be produced in the primary chamber so that the air inlet for the primary chamber can be opened already before the one for the secondary part. In this way the time for rinsing and filling can be shortened. /621

The control organs can be made hollow for cooling. Recooling, for example, is done with air, which enters the shaft and is driven through the cooling channels by means of centrifugal force (or by loader-like ribs). The cooling air is then supplied to the gas flow so that the cooling air can be exploited.

The control organs are driven by the shaft. By appropriate design of the outlet channels and of the control disks, one can insure that the combustion gases provide an impulse effect on the control organ when emerging from the primary chamber and from the secondary chamber, so that part of the flow energy is gained as drive power.

Advantages of the Method

In this combustion chamber the process operates at approximately the same volume, in contrast to the constant pressure combustion chamber, so that the combustion gases which emerge from the primary combustion chamber when the pressure drops on the average will have a higher pressure in the secondary combustion chamber than the air before combustion in the primary combustion chamber. Only the part of the gas which is pushed out during the rinsing process from the primary combustion chamber has a lower pressure than the charging air, corresponding to the pressure loss produced by the rinsing process. The pressure losses which occur because of the flow resistances, similar to the turbine with the constant pressure combustion, must also be considered. Because of the large average pressure and the higher temperature caused by the increased expansion ratio when emerging from the secondary combustion chamber, one obtains a higher performance than for the constant pressure combustion.

When the gases flow out of the primary part into the secondary part of the combustion chamber, a great deal of vorticity occurs in the secondary combustion chamber with a simultaneous recombination of the molecules dissociated in the primary combustion chamber. By appropriately shaping the overflow channel, it becomes possible to obtain essentially uniform

temperature distribution in the secondary chamber. Since the control organs are rotating disks, we have several advantages compared with valve control. The inlet and outlet cross sections are much greater than is possible using valve controls. In addition, by arranging the primary and secondary chambers in a ring, it is possible to control the working process in all the chambers in sequence using a single rotating control organ. It is then appropriate to displace the secondary parts of the chamber, arranged radially above the primary part, staggered with respect to it in the circumferential direction.

If this system is used for jet engines, it is possible to direct a partial stream of the gas mass (from the (illegible) temperature and pressure range) to the (illegible) of the control organs and of the compressor and the remainder of the gas mask can be directly supplied to the thrust nozzle.

Figure 2 shows an example of such a combustion chamber. The combustion gas first flows under a high (illegible) from the secondary combustion chamber a through the (illegible) channel b, and the cross section opened by the outlet control organ c allows the gas to flow in the ring channel d and then directly to the thrust nozzle. During the (illegible) rotation of the outlet control disk, the combustion chamber outlet opening b is closed off, after fresh gas has been partially expanded and has flowed out.

The further combustion from the combustion chamber a (illegible) through the inner overflow channel e. The overflow channels b and e are staggered in the circumferential direction so that by opening the outlet control organs, they are opened in sequence and the (illegible) can be made in a very simple way. The part of the gas mass expands from the (illegible) pressure and a low temperature (illegible) internal ring channel f and from

here it flows through the blading g and to the turbine h. The pressure drop occurs down to the compressor end pressure or (illegible) rinsing and filling. Under some conditions, the energy of the emerging gas can be used, down to a (illegible) lower pressure. Then the outlet control device (illegible) the overflow channel e in the channel system f.

Computational Treatment of the Gas Turbine Process with Intermittent Combustion

The thermodynamic process with intermittent combustion can be idealized as a compressor, primary combustion chamber, secondary combustion chamber and turbine in sequence. The primary and secondary chamber are closed off by disks. Figure 3 shows this corfiguration and will be described as follows. / 622

The compressor compresses air from the outside state T_1, p_1 to the pressure p_2 in front of the primary combustion chamber and charges the combustion chamber with fresh air with the state 2, while the inlet and outlet are open. The combustion gases still located in the combustion chamber at a pressure of p_2 are pushed out.

The compressor work is derived from the following well known formula

$$L_{iv} = \frac{1}{\eta_v} \frac{\varkappa}{\varkappa - 1} R_L T_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{\varkappa - 1}{\varkappa}} - 1 \right] \qquad (1)$$

when we assume thermally ideal gas.

Here we have:

L_{iv} Internal compressor work per kg of air,

η_v Compressor efficiency referred to isentropic combustion work,

$\varkappa$ Average isentropic exponent in temperature range T_1 to T_2.

R_L Gas constant of air

The compressor temperature T_2 follows from the relationship

$$T_2 = T_1 + \frac{L_{iv}}{c_p\Big|_{T_1}^{T_2}} \qquad (2)$$

where $c_p\Big|_{T_1}^{T_2}$ is the average specific heat in temperature range T_1 to T_2. For the additional calculations we made simplifying assumptions. When discussing the combustion chamber state 3 in the primary chamber, we assumed constant volume combustion and stoichiometric mixing ratio. Also we assumed that there was chemical equilibrium at the end of combustion, whereas it is indeed possible that at this state there will be a small amount of unburned fuel in the primary chamber, which will then burn after flowing out into the secondary chamber.

The combustion calculation was carried out using kerosine as a fuel and air as oxidizer. Therefore, 12 components can be expected in the exhaust gas, that is, O, O_2, H, H_2, OH, H_2O, CO, CO_2, N, N_2, NO and Ar.

In addition to the concentrations of the individual components, the pressure p_3 and the temperature T_3 are unknown, so that 14 nonlinear equations are applied for determining these unknowns. Five atomic balance relationships (for the O, H, C, N and Ar atoms) can be formulated as well as seven equilibrium conditions for seven chemically independent reactions, the thermal equation of state and the energy theorem, assuming constant volume.

If σ_i is the concentration or number of moles of a component i referred to the total mass of the gas mixture, β_{ij} a matrix which indicates how many moles of an element j are

contained in a component i, and if σ_{Bj} is the molar number of an element j referred to the total mass of the gas mixture, then we have the following five equations for the atomic balance:

$$\sum_{i=1}^{12} \beta_{j,i}\, \sigma_i = \sigma_{B,j} \quad \text{where } j = 1 \text{ to } 5 \tag{3}$$

The conditions for chemical equilibrium therefore result using Gibbs potential G(T,p).

For chemical equilibrium we have the following for each of the 7 chemically independent reactions among the 12 components

$$\Delta G\,(T, p) = 0 \tag{4}$$

If we introduce a Gibbs potential $G^{\circ}(T)$ referred to a standard pressure p°, we find

$$\sum_{i=1}^{12} \nu_{l,i} \left(G_i^{\circ}(T) + \Re T \ln \frac{p_i}{p^{\circ}} \right) = 0 \quad \text{where } l = 1 \text{ to } 7 \tag{5}$$

Here $\nu_{l,i}$ are the stoichiometric coefficients of the components i in the reaction l and p_i is the partial pressure of component i in the mixture.

With the assumptions of constant volume combustion in the primary chamber, we have the following for the first law

$$U_3 = U_2 + H_{Kr} \tag{6}$$

Here:

U_2 the absolute internal energy of the combustion air.

H_{Kr} the absolute enthalpy of the fuel

U_3 the absolute internal energy of the combustion gas mixture which is inchemical equilibrium.

From Equation (6) we find after substitution

$$\sum_{i=1}^{12} m_i \left[c_{v_i} \Big|_0^{T_3} T_3 + u_{0i} \right] =$$

$$= m_L \left[c_{v_L} \Big|_0^{T_3} T_3 + u_{0L} \right] + \qquad (7)$$

$$+ m_{Kr} \left[c_{Kr} T_{Kr} + h_{0\,Kr} + p_{Kr} v_{Kr} \right]$$

where u_0 and h_0 are the specific absolute internal energy and enthalpy at absolute zero, respectively. All of the quantities proportional to the amounts refer to the same standard value, such as, for example, the primary combustion chamber filling amount during a working cycle or they refer to the amount of gas passing through the primary combustion chamber per unit of time or to the mass unit of the primary combustion chamber content.

The additive constants u_{01}, u_{0L}, H_{0Kr} are numerical values which are selected so that the absolute internal energies or absolute fuel enthalpy agree with the heat of transformation of all possible reactions between the components at the given temperature variations. In this way we obtain a new system of nonlinear equations.

After combustion in the primary chamber there is a flow into the secondary chamber up to pressure equalization after the control slit has been opened.

When calculating the overflow process, it is assumed that the expansion in the primary chamber is isentropic and that chemical equilibrium prevails because the temperatures are hig in the primary chamber and for this reason the gas remains capable of performing reactions. The final state in the primary chamber is called 4 P. Since the temperature in the secondary chamber is considerably lower than the primary chamber, it is completely recombined gas in it. The pressure in the secondary chamber is calculated using the energy theorem considering the energy flowing in from the primary chamber.

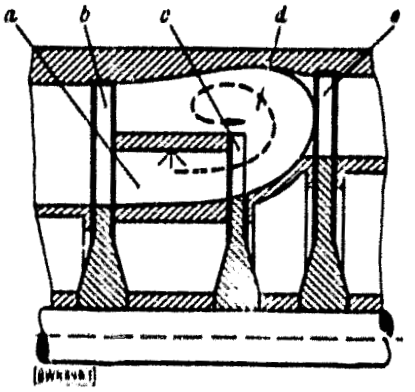

Figure 1. Schematic diagram of a gas turbine operating according to the constant volume combustion method and controlled by disks.

a- primary combustion chamber; b, c- rotating disks; d- secondary combustion chamber; e- rotating disk.

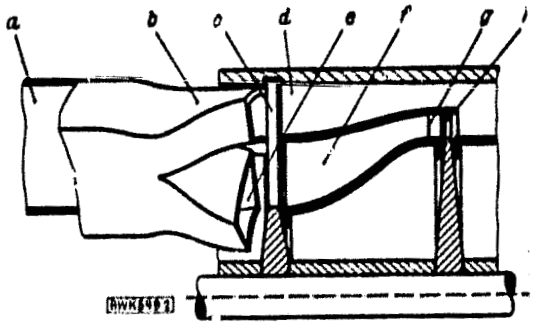

Figure 2. Application of constant volume combustion for jet engines.

a- secondary combustion chamber; b- overflow channel; c- exhaust control organ; d- ring channel; e- overflow channel; f- ring channel; g- guide blades; h- turbine.

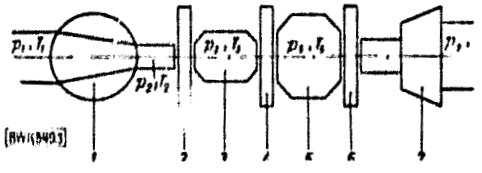

Figure 3. Diagram of a gas turbine operating according to constant volume principle.

a- compressor; b- disk; c- primary chamber; d- secondar' chamber; e- turbine

For a given pressure reduction in the primary chamber to a pressure p_{4P} we obtain all of the state variables in the primary chamber of interest by considering the equilibrium conditions and the fact that the total specific entropy is constant. We then obtain the mass flowing into the secondary chamber

$$\Delta m = m_3 \left(1 - \frac{\varrho_{4P}}{\varrho_3}\right) \tag{8}$$

where ρ_{4P} is the density of the gas mixture for the state 4P and ρ_3 is the density of the gas mixture after combustion in the primary chamber. /623

Using the energy and mass balance equations we find that the energy carried in the flow is

$$\Delta U = u_3 m_3 - u_{4P} m_3 \frac{\varrho_{4P}}{\varrho_3} \tag{9}$$

and the energy in the secondary chamber is

$$U_{4S} = (\lambda - 1) U_3 + \Delta U \tag{10}$$

where λ is the air ratio. The mass in the secondary chamber is

$$m_{4S} = (\lambda - 1) m_3 + \Delta m \tag{11}$$

Using the equations and the thermal state equation we find the pressure p_{4S} in the secondary chamber, which in general does not agree with the prescribed pressure p_{4P}, so that an iteration is required. This calculation is done in a program by means of a zero determining routine, which adapts the states 4 P and 4 S in such a way that the pressure becomes equal.

After the State 4 has been determined, which results immediately after the flowing process, the mixing process in the primary and secondary chambers is determined assuming complete recombination (state 5). According to the first law we have

$$U_5 = U_{4S} + U_{4P} \tag{12}$$

or

$$(m_{4P} + m_{4S})\left[c_{v5}\Big|_0^{T_5} T_5 + u_{05}\right] =$$
$$= m_{4S}\left[c_{v4S}\Big|_0^{T_{4S}} T_{4S} + u_{04S}\right] +$$
$$+ m_{4P}\left[c_{v4P}\Big|_0^{T_{4P}} T_{4P} + u_{04P}\right] \qquad (13)$$

Here $c_{v5}\Big|_0^{T_5}$ is the average specific heat of the gas mixture after complete recombination and after mixing

$$c_{v4P}\Big|_0^{T_{4P}}$$

is the average specific heat of the components partially dissociated and present in the primary chamber before the mixing process. $c_{v4S}\Big|_0^{T_{4S}}$ is the corresponding average specific heat for the state 4S.

State 5 corresponds to the state before the beginning of expansion in the turbine. The expansion in the turbine can be divided into two stages. First there is an expansion from state 5 to the compressor pressure state 6, $p_6 = p_2$. After calculating this pressure, the control disks open the primary and secondary chambers and a further expansion takes place to the external pressure (state 7) at the constant pressure p_2 (charge change).

During the expansion of the combustion chamber content (mass $m_5 = m_{4P} + m_{4S}$) from state 5 to the counter pressure p_7 behind the turbine, each mass element (- dm), which leaves the combustion chamber and flows through the turbine delivers the work $L_{th,x7}(dm)$ to the turbine. Here $L_{th,x7}$ is the isentropic work per unit mass for isentropic expansion of the mass particle from the instantaneous combustion chamber state T_x, P_x to the constant counterpressure $p_7 = p_1$. Since the combustion chamber state $x(T_x, p_x)$ changes continuously during the emptying process of the combustion chamber, until the compressor pressure $p_6 = p_2$ is reached and

until the state 6 (T_6, p_6) has been regulated in the combustion chamber, the quantity $L_{is,x7}$ will therefore be variable up to this time. During the subsequent rinsing process, the mass m_6 located in the combustion chamber expands at the constant combustion chamber state 6 to the constant counter pressure $p_7 = p_1$ and gives off the isentropic work

$$L_{is,67}\, m_6$$

to the turbine, see Figure 4.

The total turbine work of the work cycle therefore is

$$L_{is,V=const} = \left[\int_{m_6}^{m_5} L_{is,x7}\,(-\mathrm{d}m)\right] + L_{is,67}\, m_6 \tag{14}$$

or after a short intermediate calculation

$$L_{is,V=const} = H_5 - H_7 - V_{BK}\cdot(p_2 - p_3) \tag{15}$$

where $H_5 - H_7$ is the difference in the enthalpies of the combustion gas between states 5 and 7. V_{BK} is the total combustion chamber volume (the sum of the primary combustion chamber volume and the secondary combustion chamber volume).

The reference quantities for the quantities H_5, H_7 and V_{BK}, which are proportional to the amounts, are therefore the total filling of the primary and secondary combustion chambers during a work cycle.

We then find the following for the effective work of such a gas turbine installation per kg of gas throughput

$$L_{e,V=const} = \frac{L_{is,V=const}\,\eta_T\,\eta_{mT}}{m_5} - \frac{\lambda\, m_{L,min}}{\lambda\, m_{L,min} + 1}\cdot\frac{L_{is,V}}{\eta_V\,\eta_{mV}} \tag{16}$$

Here λ is the air ratio and $m_{L,\,min}$ is the air mass required for stoichiometric combustion.

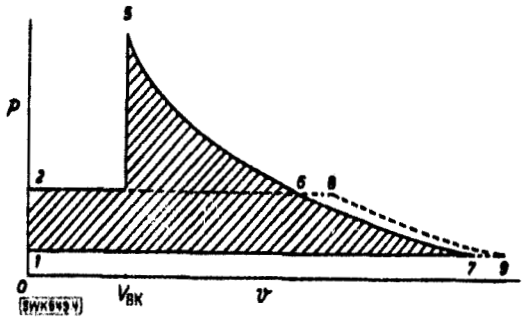

Figure 4. p,v-diagram of a gas turbine operating according to constant volume combustion method.

Determination of Average Blade Temperature

It makes sense to select the permissible running blade temperature as a reference quantity for comparison of the working method described above and a gas turbine process, because as a rule, this is the performance limiting quantity.

In the case of the gas turbine process with intermittent combustion, the pressure and the temperature in the combustion chamber changed continuously during the outflow process of the gas mixture and therefore the blade temperature also changes. Using the following computational method, it becomes possible to form a time average of the blade temperatures during a work cycle.

Since we are primarily interested in a comparison of the various working methods in this thermodynamic investigation, the running blade temperature is simplified assuming equal pressure blading, in order to limit the complexity of the calculation to a reasonable level. In order to determine the average blade temperature more accurately, it would be necessary to determine the blading design. In addition, in order to evaluate

the permissible blade temperature, it will also be necessary to know the temperature peaks which occur. If Δh_{is} is the isentropic enthalpy difference converted in the guide wheel into kinetic energy, then the outflow velocity from the guide blade can be determined according to the relationship

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$$c = \varphi \sqrt{2 \Delta h_{is}} \tag{17}$$

if the incident flow velocity is ignored. In this equation φ is the friction coefficient for characterizing expansion with friction.

For a given pressure reduction from p_5 to p_x, the corresponding enthalpy h_x can be calculated assuming isentropic state change in the combustion chamber and the gas mass m_x still remaining in the combustion chamber. If we define the degree of outflow as

$$\Phi_x = \frac{m_5 - m_x}{m_5} \tag{18}$$

then if we consider the fact that the volume is constant during expansion we find

$$\Phi_x = 1 - \frac{\varrho_x}{\varrho_5} \tag{19}$$

Here ρ is the density.

During one working cycle Φ varies within the limits between 0 and 1.

By reducing the pressure p_x in steps from p_5 to the prevailing counter pressure, we can then calculate the exit flow velocity

$$c_x = f(\Phi_x) \tag{20}$$

NUMERICAL TABLE 1. DEPENDENCE OF MIXTURE COMPOSITION IN VOLUME PERCENT ON COMPRESSOR PRESSURE RATIO IN PRIMARY CHAMBER FOR STOICHIOMETRIC COMBUSTION*

Gas	Combustion pressure ratio				
	1	4	8	12	16
O	0,1075	0,0983	0,0979	0,0993	0,1072
O_2	1,2228	1,2578	1,2478	1,2500	1,2578
H	0,0824	0,0726	0,0708	0,0709	0,0716
H_2	0,3308	0,3269	0,3269	0,3296	0,3330
OH	0,5046	0,5759	0,5957	0,6137	0,6301
H_2O	10,8021	10,8226	10,7927	10,7609	10,7213
CO	2,2430	2,2201	2,2694	2,3133	2,3556
CO_2	10,7359	10,7522	10,7102	10,6639	10,6192
N	0,0048	0,0054	0,0060	0,0066	0,0070
N_2	71,2808	71,2531	71,2078	71,1685	71,1336
NO	0,7112	0,8040	0,8751	0,9272	0,9704
Ar	1,7996	1,8000	1,7997	1,7994	1,7990

* Translator's note: Commas in numbers represent decimal points.

After the pressure has been reduced enough so that the compression pressure p_2 prevails in this combustion chamber, the combustion chamber inlet opens and the remaining outflow occurs at a constant pressure. In this way we also find a constant exit flow velocity, until the gas exchange has occurred.

The running blade temperature can be looked upon with sufficient accuracy as to be equal to the gas temperature in the blade boundary layer (see also K. Graf [4]).

Starting with Equation (20) and using the well known equations

$$w_x^2 = c_x^2 + u^2 - 2\, c_x\, u \cos\alpha_1 \tag{21}$$

and

$$T_{s,x} = T_y + \frac{\sigma\, w_x^2}{2\, c_p\,(T_y)} \tag{22}$$

it is possible to determine the blade temperature $T_{s,x}$ as a function of Φ.

In these equations w_x is the relative velocity, u is the circumferential velocity of the running wheel, α_1 is the inlet angle, σ is the recovery factor in the blade boundary layer and c_p is the specific heat of the gas mixture (average of T_y and $T_{s,x}$).

The subscript y characterizes the prevailing final state of expansion with friction in the turbine. The final temperature T_y must be determined for each computational step, i.e., for each intermediate pressure p_x located between p_5 and the counter pressure.

The average running blade temperature over a working cycle can be obtained by integrating the function $T_{s,x} = f(\Phi_x)$, as will be shown later on.

For the constant pressure process the calculation of the blade temperature is basically simple, because then we have stationary conditions. The final state of combustion (state 8) is calculated according to the assumption that we have completely recombined gas.

Numerical Procedure and Results of the Calculation

Because of the considerable computational effort for this investigation, a computer program was developed in the program language FORTRAN for the data processing installation CD 6400 of the Aachen Institute of Technology Computation Center. With this program, it is possible to calculate the gas turbine process with intermittent combustion as well as the constant pressure process for arbitrary air ratios λ, atmospheric states p_1, t_1, compressor pressure ratio π_V and turbine efficiency η_T.

The solution of the system of 14 nonlinear equations required for computing the combustion problem was done using the Newton-Raphson method which is based on linearization using a Taylor expansion. Using the Gauss elimination method, it is possible to compute correction terms from this linearized system which improves any assumed estimated system of solution. By continuous iteration up to convergence, we obtain a solution of the system of equations with a specified accuracy. For the computations made, we selected an accuracy of 0.1 $^{0}/_{00}$. The Newton-Raphson method used for the solution converges well and leads rapidly to sufficiently accurate results [6, 7].

The division of the combustion chamber into a primary and secondary part is not of interest for this basic thermodynamic calculation, because the mixing final state T_5, p_5 which results in the overall combustion chamber system can be immediately calculated from the final compression state if we start with the expansion. It is not necessary to consider the dissociation because the influence of dissociation is small at the temperatures T_5 which can become possible after mixing.

We determined the intermediate states, i.e., the state in the primary combustion chamber after combustion (state 3) and the state just after the flow process without mixing (state 4) in order to obtain information about the peak pressure and temperature loads on the primary part. Considering practical applications of the method, it makes sense because of the reaction kinetics to carry out the combustion in the primary chamber at an approximately stoichiometric mixing ratio, in order to achieve rapid ignition and a high flame propagation velocity. In the following, we will discuss and present the results of the numerical calculation.

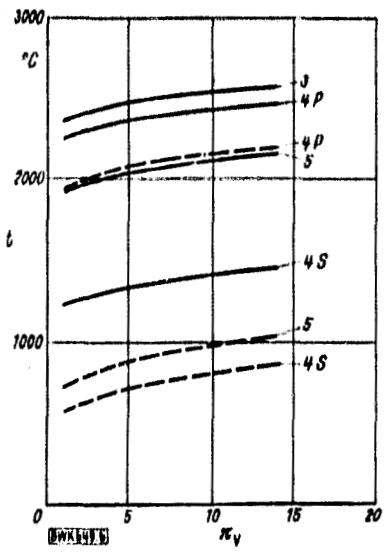

Figure 5. Temperature distribution for constant volume process as a function of compressor pressure ratio.

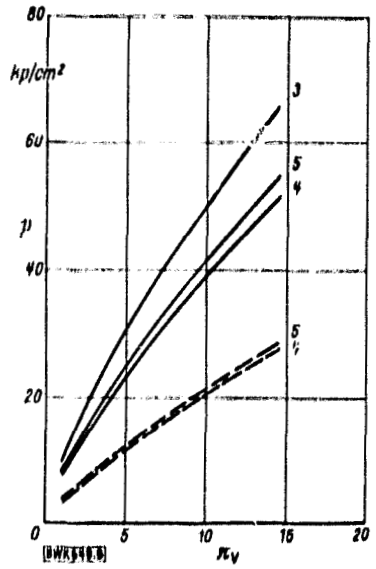

Figure 6. Pressure variation for constant volume process as a function of compressor pressure ratio.

$\lambda = l_0 \delta$
$\lambda = \delta$

3 - Primary chamber after combustion
4P - Primary chamber after expansion
4S - Secondary chamber after expansion
5 - Primary and secondary chambers after mixing.

3 - Primary chamber after combustion
4 - Primary and secondary chamber after pressure equalization
5 - Primary and secondary chamber after mixing

Numerical Table 1 shows the space fractions of the exhaust gas components which occur in the primary combustion chamber at the end of stoichiometric combustion, i.e., in chemical equilibrium as a function of the compressor pressure ratio. This state corresponds to state 3. As expected, with increasing pressure one finds a decrease in dissociation, which is compensated for by the increase in the combustion chamber temperature with pressure in the constant volume process. /625

Figures 5 and 6 show the dependence on the compressor pressure of ratio temperature and pressure at the end of combustion

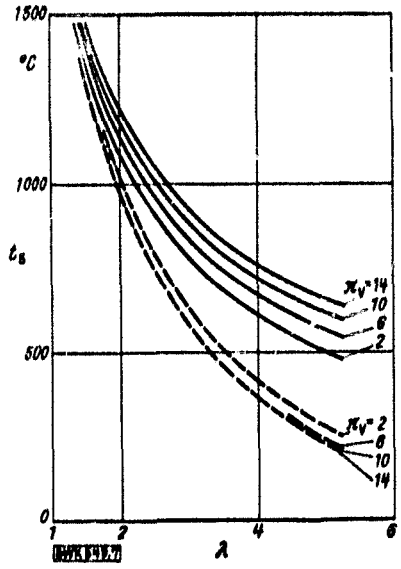

Figure 7. Average blade temperature t_s as a function of compressor pressure ratio π_V.

———— Constant volume process $\eta_T = 0{,}75$
– – – – – Constant pressure process $\eta_T = 0{,}85$

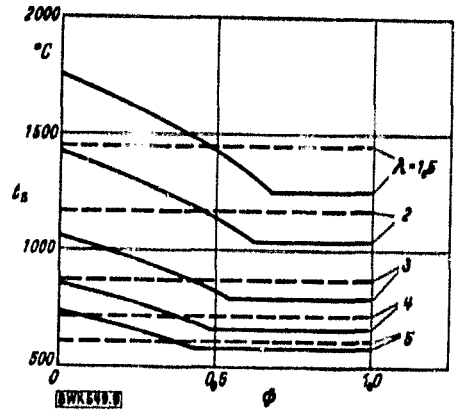

Figure 8. Blade temperature variation t_s as a function of degree of exhaust flow ϕ; compressor ratio π_V = 10, turbine efficiency η_T = 0.75

———— Temperature variation
– – – – – Average blade temperature

in the primary chamber (state 3). In addition, these figures show the further sequence of the process in the primary and secondary chambers, in terms of temperatures and pressures as a function of the compressor pressure ratio.

If we consider the temperatures which occur in the primary chamber (state 3), which were calculated for stoichiometric air ratios, it is found that the maximum temperatures which occur are on the order of 2400 to 2580° C. The corresponding pressures lie in the range between 10 to 65 kp/cm^2. These values therefore have the same order of magnitude as for the Otto motor.

Since in the present process, similar to what goes on in an Otto motor, it is possible to have rinsing with relatively

cold air, it can be assumed that the thermal load in the primary chamber will be of the same order of magnitude as for it. In addition, we must consider the fact that by selecting the combustion chamber air ratio in the primary chamber, which can be adjusted either smaller or larger than the stoichiometric ratio, it is possible to vary the peak pressures and temperatures within certain limits.

In addition, Figure 5 shows that the pressure ratio has a relatively small influence on the temperatures which occur. On the other hand, as expected, there is a strong dependence on the total air ratio. For a compressor pressure ratio, for example 10, for the state 5, we find the temperature of 980° C for an air ratio of 5, and about 2100° C for an air ratio of 1.5. Even though the temperatures shown here are peak temperatures, which drop off immediately after the outlet control organ is opened, from Figure 5 it follows that the temperature values are too high for an air ratio of 1.5. On the other hand, an air ratio of 5 results in peak temperatures which are below the values today to be considered to be permissible blade temperatures.

Figure 7 shows the average blade temperatures t_s as functions of the air ratio λ for various compressor pressure ratios π_v. The blade temperatures given here of the constant volume process are average values over a cycle, and no rinsing was considered with cold air. These give an indication of the kinds of temperatures which can occur.

If we assume an average permissible blade temperature of 900° C, then depending on the pressure ratio, the corresponding air ratio λ is about 2.5 to 3. By selecting the total air ratio we have specified the ratios of the primary and secondary chamber volumes. The calculation was carried out for a turbine efficiency of $\eta_T = 75\%$.

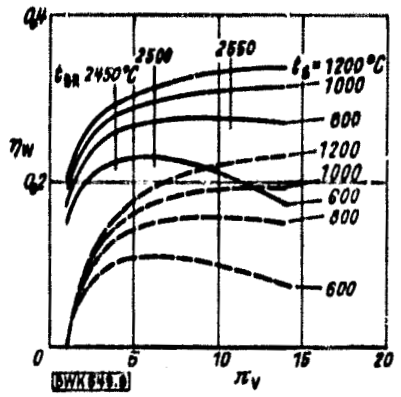

Figure 9. Economical efficiency η_W as a function of compressor pressure ratio π_V for the constant volume and constant pressure installation

——— Constant vol. process
– – – Constant pressure process

η_T = 0.75 Turbine efficiency
t_s Blade temperature
t_{BR} Temperature in primary combustion chamber

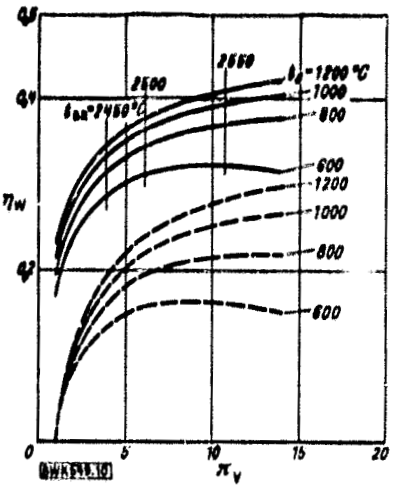

Figure 10. Economical efficiency η_W as a function of compressor pressure ratio π_V for constant volume and constant pressure installation.

——— Constant volume process
– – – Constant pressure process

η_t = 0.85 - Turbine efficiency
t_s Blade temperature

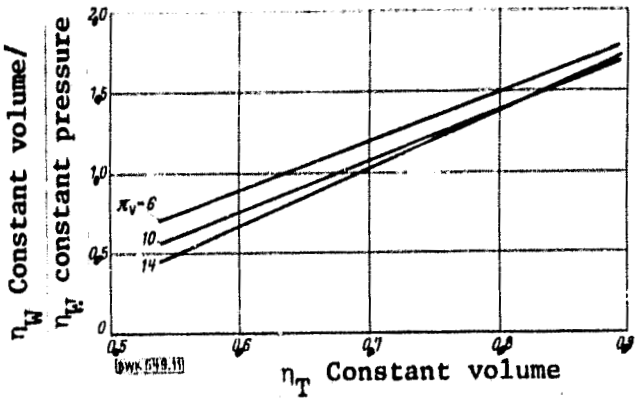

Figure 11. Ratio of economical efficiencies of the constant volume and constant pressure processes as a function of turbine efficiency of the constant volume installation.

t_s = 1,000° C Blade temperature
η_T, constant volume = 0.85 Turbine efficiency

For comparison we also show the blade temperatures for /626
a blade constant pressure process as a function of the air ratio λ and where the compressor pressure ratio is a parameter. Here we assumed a turbine efficiency of $\eta_T = 85\%$, because we must reckon with the fact that because of the changing incident flow conditions for the constant volume process, the turbine efficiency will in any case be reduced for a constant pressure process.

The dependence of the blade temperature on the degree of outflow Φ is shown in Figure 8 for a pressure ratio of 10 and a turbine efficiency of $\eta_T = 75\%$. The air ratio λ is a parameter. For comparison we also show the average blade temperatures which are done by integration.

From Figure 8 we can see that as the air ratio becomes smaller, i.e., as the temperature is increased, the fraction of the mass which flows out during the unsteady expansion process increases compared with the part which flows out during the expulsion process for constant pressure p_6.

Figures 9 and 10 show the economical efficiencies for the constant volume process and also for the constant pressure process for comparison, as functions of pressure ratio, where the average blade temperature is a parameter.

The calculation was carried out for two different turbine efficiencies η_T of 75 and 85%. As already mentioned, we will have to accept a poorer turbine efficiency for the constant volume process and for the constant pressure process. In most regions of the pressure ratio we find a considerable improvement of the economical efficiency by means of the constant volume process, if we compare the economical efficiencies for a turbine efficiency of 75% for the constant volume process and the one for a corresponding constant pressure process assuming a higher turbine efficiency

of 85%.

For a pressure ratio of for ample 10 and a permissible blade temperature of 1000°C if we consider the efficiency assumptions mentioned above, we find a relative improvement of the economical efficiency of 28.5%. If we assume that the same efficiencies can be achieved for the constant volume process as for the constant pressure process, then this would result in an improvement of 57%. The corresponding absolute values are about 39.5% and therefore are of the same order of magnitude as for a diesel motor.

Since the constant volume process can operate theoretically without precompression, in the limiting case of a pressure ratio of 1 we obtain a finite efficiency, but for the constant pressure process the efficiency becomes 0. This is also evident from Figures 9 and 10.

The influence of the turbine efficiency can be seen in Figure 11. This figure shows the ratio of the economical efficiency for the constant volume process, referred to the economical efficiency for a constant pressure process assuming that the turbine efficiency of the constant pressure process is 85% and remains constant. The calculation was carried out for a blade temperature of 1000°C.

As Figure 11 shows, for a compressive pressure ratio of 10 we also find a considerable improvement when the turbine efficiency of a constant volume installation is smaller than 75%. Even for a turbine efficiency of the constant volume process of only 67% we still find the same economical efficiency as for the constant pressure installation. For decreasing compressor pressure ratios, these values are displaced considerably in favor of the constant volume installation.

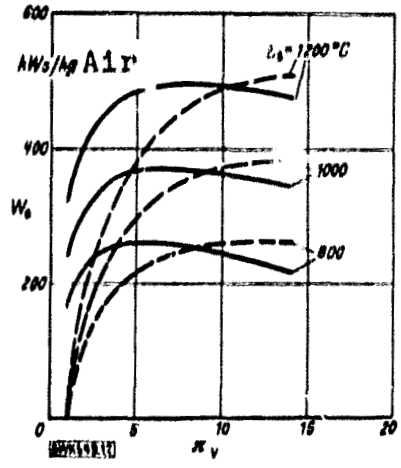

Figure 12. Specific power W_e of constant volume and constant pressure process as functions of compressor pressure ratio.

——— Constant volume process: $\eta_T = 0.75$
– – – Constant pressure process: $\eta_T = 0.85$
t_s Blade temperature

Figure 12 shows the specific power referred to 1 kg of air throughput for various permissible blade temperatures as a function of the pressure ratio. In the calculation we again assumed a turbine efficiency of 85% for the constant pressure process and an efficiency of only 75% for the constant volume process. The figure shows that at the higher pressure ratios, depending on the permissible blade temperatures, the constant pressure process will result in higher performance. On the other hand at low pressure ratios, the constant volume process offers advantages, also with respect to the specific power, compared with the constant pressure process.

SUMMARY

The present paper contains a thermodynamic investigation of a gas turbine process with controlled constant volume combustion and a comparable constant pressure process. The calculation was

mode for various compressor pressure ratios, blade temperatures and with consideration of dissociation in the combustion (illegible). In addition, the influence of the turbine efficiency was shown for the constant volume and constant pressure processes. However, due to the alternating incident flow in the case of the constant volume process, one must accept a poorer effeciency for the constant pressure process.

One important result of the calculation is that for the same permissible average blade temperature the constant volume process offers significant advantages compared with the constant pressure process within wide limits of the pressure ratio and the turbine efficiency. For the example of a pressure ratio of 10 and a permissible blade temperature of 1000°C, we find an increase in the economical efficiency of 57%, assuming that the turbine efficiencies are the same for both processes. This result holds for the assumption that the turbine efficiecy is 85% for both the constant pressure and the constant volume process.

Even for considerably reduced turbine efficiencies for the constant volume process, one also finds a considerable improvement of the economical efficiency. Only for a turbine efficiency of 67% for the constant volume process are the economical efficiencies the same in both cases. As the pressure ratio is decreased the conditions change noticeably in favor of the constant volume installation, which as is well known, leads to finite efficiencies even without precompression.

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